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Appl. No. 10/761,865
Applicant Koneda et al.
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Examiner Chang, Ching
Docket No. 81044248
Customer No. 33066



SUPPLEMENTAL DECLARATION UNDER
37 CFR 1.131

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T.C./A.U. 3748
Commissioner for
Patents P.O. Box 1450
Alexandria, VA 22313-1450

We, James Ervin, Thomas Megli and Philip Koneda, state that:

1. We are joint inventors of the above-identified patent application;
2. We made an invention claimed in the above-identified patent application in the United States prior to February 4, 2003.
3. In support thereof, we submit the following evidence:
 - A. A Document (see attached) created by the above-

identified James Ervin entitled "DISCLOSURE OF VALVE ACTUATOR USING A HYDRAULIC DISPLACEMENT AMPLIFIER" by James D. Ervin, Thomas Megli and Philip Koneda, the date on said Document having been removed but such date being prior to February 4, 2003, said Document being attached as attached EXHIBIT A;

B. Said Document in EXHIBIT A shows on page 3 in Figure 3 a hydraulic lever similar to FIG. 3 of the above-identified patent application.

C. Said Document in EXHIBIT A shows on page 3 in Figure 3 a hydraulic lever similar to FIG. 3 of the above-identified patent application more particularly:

An electronic valve actuator, comprising:

an electromagnet;

an armature disposed adjacent to the

electromagnet;

a fluid-containing chamber having:

a first piston providing a first wall portion of the chamber; and

a second piston providing a second wall portion of the chamber, the first wall portion having a greater surface area than the surface area of the second wall portion; and

wherein the first piston is coupled to the armature and the second piston is coupled to a valve.

D. On or about January 01, 2003, it was, as an initial first step in reducing the electronic valve actuator having a fluid-containing chamber having a first piston providing a first wall

portion of the chamber; and a second piston providing a second wall portion of the chamber, the first wall portion having a greater surface area than the surface area of the second wall portion to initially fabricate the valve shown in Figure 3 of Exhibit A with the two pistons forming portions of the walls of the chamber having equal surface areas (i.e., a 1:1 ratio device). More particularly, we chose to make a 1:1 ratio device as a matter of convenience because it allowed us to utilize our pre-existing hardware and supporting test fixtures. Although this prototype was realized at 1:1, ALL of our modeling and analysis documentation was focused on characterizing behavior on ratios greater than 1:1. To put it another way, it was our intention to fabricate an electronic valve actuator having a fluid-containing chamber having a first piston providing a first wall portion of the chamber; and a second piston providing a second wall portion of the chamber, the first wall portion having a greater surface area than the surface area of the second wall portion, as described and shown in Figure 3 of Exhibit A after the initial fabrication of the 1:1 ratio device. For our purposes, the real value of hardware is to validate the relationships being modeled and the 1:1 hardware was convenient for that purpose (confirm leakage 1 tolerance / viscous relationships, etc.). We then rely on the models to allow us to predict behavior at the ratio condition other than 1:1 that is most appropriate for a given application, package space, etc.

E. The EXHIBITS accompany our previous filed

declaration dated July 19, 2006 provide documentation of our activity towards first building a 1: 1 ratio device as a first step in building the electronic valve actuator a fluid-containing chamber having a first piston providing a first wall portion of the chamber; and a second piston providing a second wall portion of the chamber, the first wall portion having a greater surface area than the surface area of the second wall portion.

F. Attached are clearer copies of Exhibits A, E and N filed with our previous filed declaration dated July 19, 2006.

G. Exhibit E clearly shows an actuator having a fluid-containing chamber having a first piston providing a first wall portion of the chamber; and a second piston providing a second wall portion of the chamber, the first wall portion having a greater surface area than the surface area of the second wall portion. As noted the date on the Exhibit E is April 25, 2003.

4. All statements made herein of my own knowledge are true and that all statements made on information and belief are believed to be true.

5. We understand and have been advised that willful false statements and the like made herein are punishable by fine or imprisonment, or both (18 U.S.C. 1001) and may jeopardize the validity of the application or any patent issuing thereon.

Date: 1/18/07 JE

J-E
James Ervin

Date: 1-18-2007

T. Megli
Thomas Megli

Date: 1-18- 2007

P. Koneda
Philip Koneda

DISCLOSURE OF VALVE ACTUATOR USING A HYDRAULIC DISPLACEMENT AMPLIFIER

James D. Ervin, Thomas Megli, Philip Koneda

Introduction

One common approach to controlled valve actuation is to use two electromagnets to toggle an armature connected to a valve between an open or closed position where it is held, while a pair of springs is used to force the valve to move (oscillate) to the other state (Fig. 1). A common shortfall to such a design is that the magnets must generate force over a travel length equal to the valve stroke. At points of travel where the air gap is large, more current is required to achieve a given force resulting in a corresponding increase in power consumption. Conversely, the peak force that can be generated for a given current is reduced as the air gap increases, which effectively reduces the authority to control the valve motion.

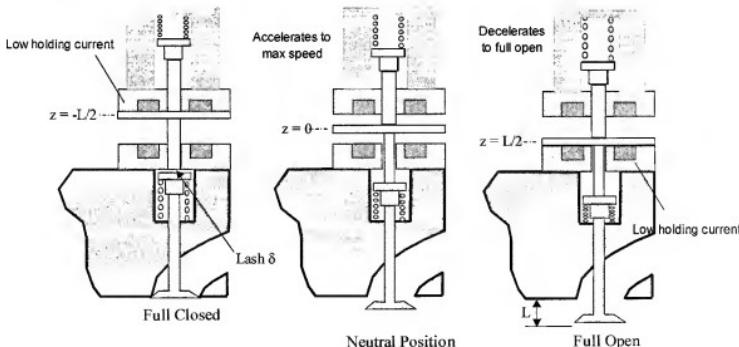


Figure 1: Conventional linear direct acting spring-mass oscillator

In contrast to the direct acting mass-oscillator approach, a new actuation architecture is proposed here which utilizes a hydraulic lever amplify the motion of a magnetic

armature to achieve a desired valve displacement, thus reducing the effective air gap.

This disclosure describes the potential claims regarding this new architecture:

- Use of a hydraulic lever to minimize variation in air gap.
- Incorporation of passive hydraulic lash adjustment in the hydraulic lever mechanism.
- Incorporation of passive hydraulic damping in the hydraulic lever mechanism to potentially enable open loop control.
- Incorporation of a bypass valve to tune the damper response and allow for initialization of the system at startup.
- Ability to flexibly package the system to meet package requirements.

Prior art

As a departure from the direct acting systems, there is at least one example of a lever actuated system: LSP Innovative Automotive Systems of Munich, Germany has developed an actuator that uses a mechanical lever to amplify the travel of the armature and reduce the effective air gap. In contrast to the hydraulic system proposed here, the mechanical lever does not address issues of passive lash management or passive damping and must be designed within specific packaging rules. As a result of the reduced gap, the LSP lever system does improve the control authority through the stroke and improves the power consumption relative to conventional linear oscillators.

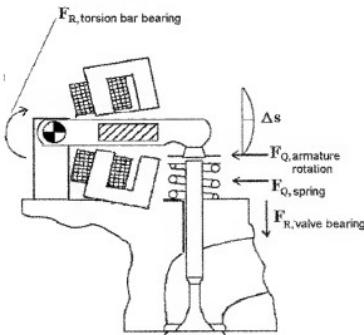


Figure 2: LSP lever acting spring-mass oscillator

Overview of the hydraulic amplifier lever oscillator

As the subject of this disclosure, a hydraulic lever is proposed as an alternative means of amplifying armature displacement. This has the effect of reducing the armature travel that is required to achieve a desired valve displacement and, in turn, reduces the effective air gap. In one proposed arrangement (Fig. 3), a first hydraulic piston is attached to an armature and biased with a first spring to be normally held in a downward position while a second piston is attached to a valve and biased with a second spring in a normally upward position.

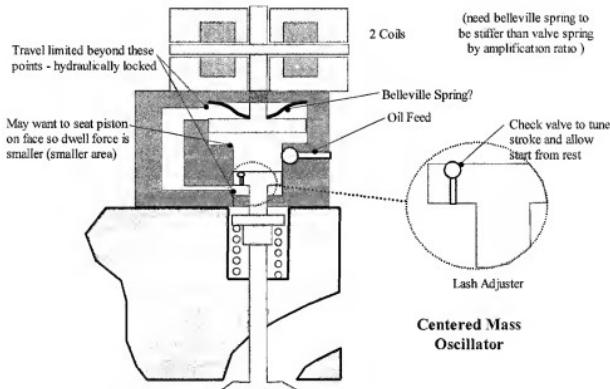


Figure 3: Hydraulic lever acting spring-mass oscillator

During a startup sequence, an upper coil is used to pull the armature upward. This creates higher pressure in the outer cavity than the central cavity to open a check valve mounted on the second piston, allow fluid transfer from the outer cavity to the central cavity, and compress the first spring.

Following this initialization process, the upper coil is de-energized, causing the first spring to move the armature and first piston downward. This increases the pressure in the central cavity between the piston faces, decreases the pressure in the outer cavity, and closes the check valve. The pressure difference across the second piston causes it to move downward and compress the second spring. At some time during this process, the lower coil is energized to continue compressing the second spring until the second piston strokes beyond the point where it can communicate oil with the outer cavity. At this

time, the second piston becomes hydraulically locked, travel stops, and the valve is held in the open position. Alternately, the travel limits could be regulated by designing the first piston to strike a shoulder in the central cavity or by driving the armature to land on the lower coil. are expected to be that is controlled by opposing coils.

Conversely, the lower coil can be de-energized and the upper coil can be energized to reverse the process and close the valve. It is expected that the pressure in the central cavity will be greater than that of the outer cavity during this event so that the check valve remains closed. This expectation requires that the force from the upper coil minus the force from the upper spring be less than the force of the second spring at all times and can be ensured by designing the preload of the second spring appropriately.

Lash Adjustment

The use of hydraulic lash adjusters for controlling tolerance stackup and thermal growth in valvetrains is well established. Each cycle following the seating of the valve, a controlled leak (orifice) from an oil lube reservoir is used to charge a hydraulic chamber to eliminate the clearance between the valve stem and the cam profile. In the case of EVA, this lash adjuster would be positioned between the valve stem and the actuation source. When acted upon by the cam, the oil becomes trapped in the cavity (can't leak out fast enough) and pushes on the valve with essentially no lash. The key to providing lash adjustment in this manner is designing the controlled leak to be slow enough to allow only a minimal change in length during the valve stroking process while being fast enough to account for the rate of length change due to thermal growth. It must also operate across a wide range of viscosity as the oil temperature.

Referencing the hydraulic lever implementation shown in Fig. 3, hydraulic lash control could be readily achieved by designing the appropriate clearance between the support body and the upper and lower pistons. In a typical sequence of operation, the lower piston would stroke upwards until the valve became seated. The upper piston would then continue stroking until its travel became limited by the seating of the armature or by a closing off of the channel that communicated oil to the underside of the lower piston (hydraulic locking). During this event, a differential pressure would develop across the upper piston, causing fluid to flow into the central cavity through the check valve and leak between the perimeter of the upper piston and the support body to resolve

the volume displaced. Having charged the cavity, the actuation source would be able to begin opening the valve with no lash.

During an opening event, downward motion of the upper piston would cause the lower piston to stroke downward, compressing the valve spring and creating a differential pressure across the lower piston. As a consequence of the differential pressure, fluid would flow out of the central cavity, resulting in a lower net stroke of the lower piston than for the upper piston. Such lost stroke is actually desired to account for valve growth due to thermal effects, where the loss of stroke (leakage) is ideally designed to be greater than the maximum thermal growth that can occur during a given cycle.

As a tradeoff during the valve closing event, the length lost during opening would result in the valve landing before the upper piston had finished stroking. With the natural coupling of position and velocity for the upper and lower pistons, it is advantageous to design the leakage to be as small as possible so that the travel of the two pistons is nearly the same. Thus if one is landed gracefully, both will, leading to simpler control.

Passive Damping

Taking advantage of the hydraulic architecture, it is also simple to incorporate passive damping into the actuator. As an example (Fig. 4), the travel of the pistons could be damped at the travel extremes by having the pistons close off a port, stopping the communication of oil to the rest of the circuit. At this point the system would be hydraulically locked though a conservation of volume. A check valve could be incorporated into the overtravel portion of the cavity to facilitate the release event. In a further refinement, the shape of the port could be tailored to provide a desired level of damping as a function of piston travel (ref. docket 201-1552 Variable Area Damper), as suggested in Fig. 4. Other implementations could include a ring extending from the piston and engaging mating cavity on the support (ram damper).

Passive, velocity-dependent damping offers significant advantages over active EVA control:

1. Reduces or eliminates the need for high speed, complex position and current feedback control of the EVA solenoids -- This complex control is presently required to achieve soft valve seating velocity for present actuator systems. The feedback control requires a high-speed (approximately 10-20kHz) control loop

frequencies) computer, a position sensor for every engine valve/EMVA assembly. The control algorithms required for soft-landing are highly nonlinear, and require complex structures, such as adaptive or iterative learning control schemes to compensate for changes in actuator and valve characteristics over the life of the engine.

2. Improves system robustness and repeatability – The damper reacts to remove nearly all of the armature kinetic energy near the end of a valve transition; therefore, the damper achieves low contact velocities for a wide range of solenoid voltage control inputs. This robustness demonstrates that the damper will compensate for changes in operating characteristics due to manufacturing variability, engine wear, fluctuations in vehicle supply voltage, and changes in gas flow force disturbances.

As a drawback, such passive damping does dissipate energy and will increase power consumption relative to the undamped case. It should be noted however that active control methods to manage impact events, such as lash take-up and landing, increase power consumption as well. The magnitudes of these effects are still under investigation.

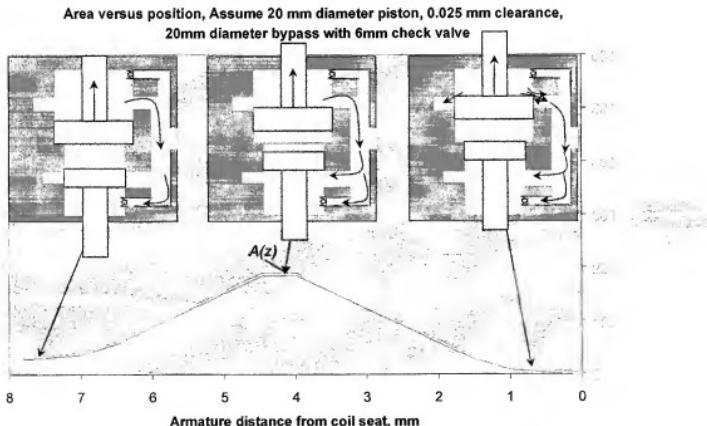


Figure 4: Example Implementation of Passive Damper

Vibration Cancellation

For mass-oscillating EVA systems, the typical valve acceleration and combined valve/armature mass is greater than for conventional cam designs leading to increased engine vibration. In an effort to minimize this effect, a design variant of the hydraulic lever is proposed where the upper port of the upper piston communicates with the upper port of the lower piston and the lower ports are similarly connected. This arrangement forces the motion of the two pistons in opposite directions, so that the inertial loads of each tend to cancel according resulting in a net force:

$$F_{net} = m_1 a_1 - m_2 a_2$$

where m_1 and m_2 are the masses of the upper and lower pistons and a_1 and a_2 are their respective accelerations.

Considering an example using armature and valve masses from conventional linear oscillator technology, a peak force of 630N would be produced by a 400g accel acting on the 0.090Kg upper mass (armature / stem / spring) and 0.073Kg lower mass (valve / keeper / retainers / spring). If the masses were driven in opposition by a hydraulic amplifier (1:1 amplification), the net force would instead be 66N, a tenfold reduction. If a larger amplification were considered, experience with the LSP lever system suggests that the armature mass would need to increase in keeping with the lever ratio while the associated acceleration would decrease.

Donut and hole opposed

Variable orifice damping

Find armature (upper) mass as a function of design parameters and valve (lower) mass:

$$\Delta t_{trans} = \pi \sqrt{\left(\frac{L \times SF \times TR}{2 \times FPAM} \right) \left(\frac{M_v}{M_a} + \frac{1}{TR^2} \right)}$$

$$\frac{M_v}{M_a} = \frac{2 \times FPAM \times (\Delta t_{trans})^2}{L \times SF \times \pi^2 \times TR} - \frac{1}{TR^2} = \frac{2 \times FPAM \times (\Delta t_{trans})^2 \times TR \times \text{sign}(TR) - L \times SF \times \pi^2}{L \times SF \times \pi^2 \times TR^2}$$

$$M_a = M_v \left(L \times SF \times \pi^2 \times TR^2 \right) \left(2 \times FPAM \times (\Delta t_{trans})^2 \times TR \times \text{sign}(TR) - L \times SF \times \pi^2 \right)^{-1}$$

Significant mass ratio relationships:

$$\frac{M_a}{M_v} = \frac{\left(L \times SF \times \pi^2 \times TR^2 \right)}{\left(2 \times FPAM \times (\Delta t_{trans})^2 \times TR \times \text{sign}(TR) \right) - \left(L \times SF \times \pi^2 \right)} = \frac{A}{B - C}$$

- 1) Lowest TR that can be applied for a given transition time (TR_{\min}):
 Armature mass goes to ∞ when TR satisfies the relationship:

$$B - C = 0 \Rightarrow TR \times \text{sign}(TR) = \frac{L \times SF \times \pi^2}{2 \times FPAM \times (\Delta t_{trans})^2}$$

Armature mass also goes to ∞ when TR goes to ∞ .

- 2) Find TR which produces the minimum armature mass (TR_{opt}):
 Armature mass is minimum when:

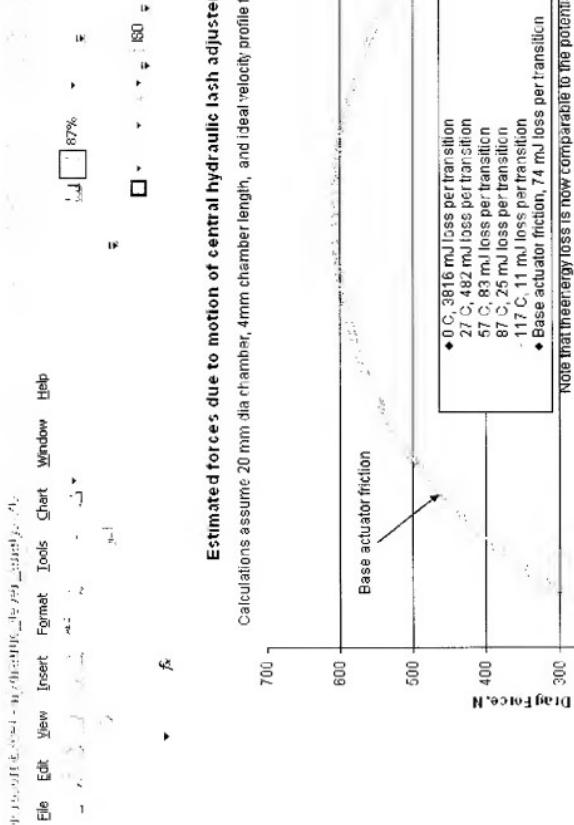
$$\frac{\partial M_a}{\partial (TR)} = 0 = M_v \left(\frac{2(L \times SF \times \pi^2 \times TR)}{(2 \times FPAM \times (\Delta t_{trans})^2 \times TR \times \text{sign}(TR) - L \times SF \times \pi^2)} - \frac{(2 \times FPAM \times (\Delta t_{trans})^2 \times \text{sign}(TR)) (L \times SF \times \pi^2 \times TR^2)}{(2 \times FPAM \times (\Delta t_{trans})^2 \times TR \times \text{sign}(TR) - L \times SF \times \pi^2)^2} \right)$$

$$\frac{\partial M_a}{\partial (TR)} = 0 = M_v \left(\frac{(2 \times L \times SF \times \pi^2 \times TR) (2 \times FPAM \times (\Delta t_{trans})^2 \times TR \times \text{sign}(TR) - L \times SF \times \pi^2) - (FPAM \times (\Delta t_{trans})^2 \times \text{sign}(TR) \times TR)}{(2 \times FPAM \times (\Delta t_{trans})^2 \times TR \times \text{sign}(TR) - L \times SF \times \pi^2)^2} \right)$$

$$\frac{\partial M_a}{\partial (TR)} = 0 \Rightarrow TR \times \text{sign}(TR) = \frac{L \times SF \times \pi^2}{FPAM \times (\Delta t_{trans})^2}$$

Notice that TR_{opt} is exactly twice the value of TR_{\min} .

- 3) As TR is reduced from the TR for min arm to the lowest applicable TR value, the effective mass is growing like
- 4) As TR approaches ∞ from the TR for minimum armature mass, the armature mass increases because more actuator force is required to deliver the same force to the valve.



cold conditions. The motion on will now be highly coupled to the drag forces, which will substantially reduce the velocity increase the transition time. The ideal velocity profile will overestimate the loss but the loss will still be comparatively large. This points to potential problems with actuator friction under cold engine conditions.

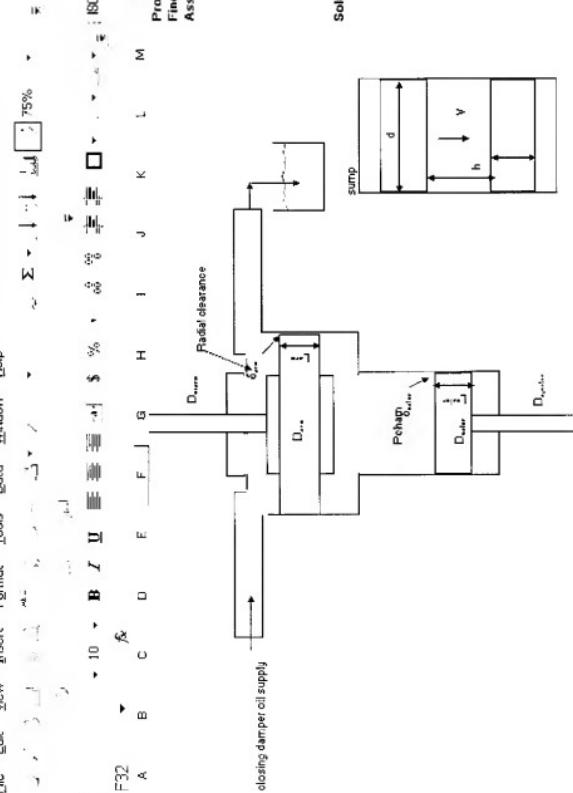


1. $\frac{1}{2} \rho_1 v_1^2 (1 + \epsilon_1) [1 - (1 + \epsilon_1)^{-1} \cdot \frac{D_{out}}{D_{in}}] = \frac{1}{2} \rho_2 v_2^2 (1 + \epsilon_2) [1 - (1 + \epsilon_2)^{-1} \cdot \frac{D_{out}}{D_{in}}]$

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Solution Method:

1. Compute Reynolds number from average velocity, viscosity, and pipe diameter.

$$Re = \frac{\rho V d}{\mu}$$

2. Calculate Darcy friction factor from Hazen-Williams equation.

$$\frac{1}{4} f = -2.0 \log \left(\frac{e (d)}{3.7} + \frac{2.51}{Re^{0.9}} \right)$$

3. Compute Shear stress from Darcy friction factor.

$$\tau_s = \frac{8 \pi}{\rho d} f u^2$$

4. Calculate drag force by integrating shear stress over chamber width.

$$F_d = \tau_s \cdot 2h$$

5. Integrate drag force over position to calculate drag forces.

$$F_d = \int_{-L/2}^{L/2} F_d(x) dx$$

Exhibit N

Exhibit N

17

Ervin iEVA program activity from 2/27-3/10/03

Ervin iEVA program activity from 3/10-5/15/03

Ervin iEVA program activity from 5/15-6/23/03

Ervin iEVA program activity from 6/23-6/30/03

iEVA program activity from 6/30-7/15/03

Ervin iEVA program activity from 7/15-7/28/03

Ervin iEVA program activity from 7/28-9/9/03

Ervin iEVA program activity from 9/10-10/8/03

Ervin iEVA program activity from 10/8-10/21/03

Ervin iEVA program activity from 10/21-11/26/03

Ervin iEVA program activity from 11/24-1/20/04